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Dennis N. Greeley  
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DESIGN AND THERMAL PERFORMANCE TESTING  
OF A HEAT PIPE FLAT PLATE SOLAR COLLECTOR

BY

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B.S.E., Florida Technological University, 1976

THESIS

Submitted in partial fulfillment of the requirements  
for the degree of  
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of Florida Technological University

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1977



## ABSTRACT

This thesis investigates the use of heat pipes for removing heat from flat plate solar collectors and transferring the energy to air in a space heating system. Heat pipes are passive devices which are very efficient at transporting heat energy. They operate using a closed evaporation-condensation cycle.

This thesis presents the fundamentals of flat plate collectors, heat pipes, and a set of parameters for evaluating a heat pipe flat plate collector. Also included are the results of this project and the recommendations for improvements.



## ACKNOWLEDGEMENTS

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## LIST OF SYMBOLS

A	= wick free flow area
$A_c$	= collector area
$A_e$	= edge area
$A_f$	= total heat transfer area
B	= pipe spacing
$C_b$	= bond conductance
CFM	= cubic feet per minute
D	= capillary hydraulic diameter
F	= fin efficiency
$F'$	= collector efficiency factor
$F''$	= flow factor
$F_c$	= incident energy absorbed by collector plate
$F_e$	= effective transmittance-absorptivity product
G	= mass flow rate per unit of collector area
H	= rate of incident radiation
I	= amount of energy incident on the collector
$I_D$	= diode current
$I_r$	= reverse diode current
K	= extinction coefficient for the cover plate
L	= path length of light through the cover plate
$L^*$	= capillary length



$L_i$	= insulation thickness
$N$	= a diode constant
$Q_L$	= rate of energy loss from the collector
$Q_{max}$	= maximum heat transfer rate for a heat pipe
$Q_s$	= rate of energy storage in the collector
$Q_u$	= rate of useful heat transfer from the collector to the working fluid
$R$	= tilt factor
$R_h$	= thermal resistance of the heat pipe to air interface
$S$	= amount of energy incident on the collector plate
$T$	= temperature
$T_a$	= ambient air temperature
$T_b$	= collector plate temperature
$T_i$	= inlet fluid temperature
$U_b$	= back loss coefficient
$U_e$	= edge loss coefficient
$U_L$	= overall loss coefficient
$U_o$	= overall heat transfer coefficient between the ambient air and the heat removal fluid
$U_t$	= top loss coefficient
$V_D$	= diode voltage
$V_f$	= fin face velocity
$b$	= bond length
$b^*$	= capillary geometry constant
$c_p$	= specific heat of the collection fluid



$d$	= inside pipe diameter
$d_o$	= outside pipe diameter
$dt$	= time period
$h$	= evaporation film coefficient
$h^*$	= latent heat of vaporization of the working fluid
$h_f$	= convection coefficient, for fin
$h_{p-c}$	= convection coefficient, collector plate to cover
$h_{r,c-s}$	= radiation coefficient, cover to sky
$h_{r,p-c}$	= radiation coefficient, collector plate to cover
$h_w$	= convection coefficient, wind
$k$	= Boltzman's constant
$k^*$	= collector plate thermal conductivity
$k'$	= bond thermal conductivity
$k_i$	= insulation thermal conductivity
$n$	= refractive index of the cover plate
$q$	= electron charge
$q'$	= amount of energy transferred to the pipe from the collector plate
$q'_u$	= total useful energy gain per unit of collector length
$b$	= subscript for beam radiation
$d$	= subscript for diffuse radiation
$\gamma$	= average bond thickness
$\delta$	= collector plate thickness
$\eta$	= collector efficiency
$\theta_i$	= angle of incidence



$\theta_r$	= angle of refraction
$\nu$	= kinematic viscosity
$\rho$	= reflectivity
$\sigma$	= surface tension
$\tau$	= transmittance
$(\tau\alpha)$	= transmittance-absorptance product



## CHAPTER I

### INTRODUCTION

The purpose of this project was to research the use of heat pipes in flat plate solar collectors. Heat pipes are passive devices that have the equivalent of a high thermal conductance. Because of this, they can be used to remove heat from a flat plate collector and transfer the energy to the working fluid of a distribution system. There are several advantages to this approach over the conventional flat plate collector. They are (1):

1. the heat pipe is a closed system so corrosion problems can be eliminated by careful selection of materials
2. proper selection of materials can also give protection from freezing
3. heat pipes are passive devices so no pumps or fans are required to move the working fluid through the collector
4. a heat pipe collector can be designed and fabricated as modules which would ease installation and simplify maintenance
5. each heat pipe, being a closed system, would mean that the failure of an individual pipe would not cause the entire system to fail
6. if wickless heat pipes, with the return of the condensate



dependent upon gravity, are used, the heat pipes will act as thermal diodes

7. use of wickless heat pipes can increase the average collection rate during periods of intermittent insolation by preventing the reradiation of collected energy back to the environment by the collector
8. use of heat pipes would allow the design of collector plate geometries that would give higher daily collection rates

The price paid for these advantages are added thermal resistances that are introduced, mainly the interface between the heat pipe and the distribution system's working fluid.

A basic understanding of heat pipes is required to understand their application to solar collectors. A heat pipe, as shown in Figure 1, works best if it is long and thin, such as a cylinder with a large length to diameter ratio (2). A conventional heat pipe has a container lined with a wicking that is saturated with liquid. The pressure within the container is close to the vapor pressure of the fluid for specified operating temperature range. When heat is added to the pipe at the evaporator area, the liquid vaporizes creating a pressure gradient in the container. This pressure gradient causes the vapor to flow to the lower pressure region, the condenser, where it gives up its heat as it condenses. The condensate is transported back to the evaporator through the wick by a capillary pumping action.

The movement of the liquid through the wick is caused by a



difference in the contact angle and the meniscus radius of curvature, in the evaporator and condenser areas. The evaporation of liquid decreases the contact angle while condensation increases this angle. Because of this operation, it is necessary that the fluid have a contact angle of less than  $\frac{\pi}{2}$  to insure wetting of the wick (2).

The best position for the wick is against the inside wall of the pipe. This allows the radial heat transfer to occur through the medium of highest conductance to minimize radial temperature differences. This also maximizes the hydraulic diameter of the vapor space. The wick can be a woven cloth, felt, or even grooves cut in the wall of the pipe. Whatever is used, it can be characterized by its permeability, liquid volume fraction, and mean pore size (2).

Another approach is to not use a wick to transport the liquid back to the evaporator. In this method the condenser is higher than the evaporator. This elevation of the condenser allows the fluid to flow back to evaporator by gravity. In some cases a wick may be used in the evaporator to assure a uniform supply of working fluid around the perimeter of the heat pipe (2).

To minimize the radial temperature gradient, the walls of the container should be as thin as possible. To do this, the working pressure inside the pipe should be near the ambient pressure so the container's wall can be as thin as possible (2).

Since the heat transfer is mainly due to the latent heat of vaporization of the liquid used and the rate of energy, transfer is



limited by how quickly the working fluid can be returned to the evaporator. The maximum heat transfer rate,  $Q_{\max}$ , can be expressed as

$$Q_{\max} = \frac{\sigma A D h^*}{b^* v L^*} \quad (1)$$

where

$\sigma$  = the surface tension of the working fluid

$v$  = the kinematic viscosity of the working fluid

$b^*$  = a dimensionless constant for the capillary geometry

$A$  = the free flow area in the wick

$D$  = the hydraulic diameter of an individual capillary

$L^*$  = the capillary length

$h^*$  = the latent heat of vaporization of the working fluid

All of this indicates that the working fluid should be carefully chosen to have good thermophysical properties and be compatible with the container material. This is dealt with in greater detail in reference 3.

#### FLAT PLATE COLLECTORS

The flat plate solar collector is a means of converting radiant energy from the sun into useful thermal energy. In this type of collector, the area of the absorber is equal to the area intercepting the radiant energy. The collector usually has a



transparent cover over the radiation absorbing collector plate which has its back and edges insulated to minimize the loss of the collected energy. The collector normally has some means of transferring the heat to a working fluid. Since the maximum incident radiation, within the wavelength range from  $0.3 \mu\text{m}$  to  $3.0 \mu\text{m}$  is approximately  $350 \frac{\text{BTU}}{\text{ft}^2\text{-hr}}$  and is not concentrated, the collector is normally designed to operate at temperatures up to about  $180^\circ\text{F}$  above ambient (4). However, this moderate temperature range is of interest for domestic water heating and space heating.

Without the focusing of the incoming energy, both beam and diffuse radiation can be used. This is an advantage over concentrating types of collectors that do not capture the diffuse radiation which can be a substantial fraction of the total radiation on a partly cloudy day (5). Unlike the concentrating type of collector the flat plate collector has the advantage of not requiring a tracking mechanism. This results in the flat plate collector being relatively simple as compared to the concentrating type collector.

The rate at which useful energy can be collected depends upon several related factors. These are

1. the amount of radiant energy available
2. the amount of energy passed through the cover plate and absorbed by the absorbing plate
3. the rate at which energy is lost through reradiation, conduction and convection
4. the efficiency of transferring the absorbed energy into the



working fluid

How well any collector should perform can be determined by writing an energy balance for the collector. This can be written as (4)

$$A_c \left[ [HR(\tau\alpha)]_b + [HR(\tau\alpha)]_d \right] = Q_u + Q_s + Q_L \quad (2)$$

where

$A_c$  = area of collector

$H$  = rate of incident radiation (beam or diffuse)

$R$  = factor to convert beam or diffuse radiation  
to that on the plane of the collector

$(\tau\alpha)$  = transmittance-absorptance product for the  
cover system

$b$  = subscript indicating beam radiation

$d$  = subscript indicating diffuse radiation

$Q_u$  = rate of useful heat transfer to the working  
fluid

$Q_L$  = rate of energy loss from the collector

$Q_s$  = rate of energy storage in the collector

In analyzing any collector one of the first components to be considered is the transparent cover. The main purpose of the cover is to minimize convective heat losses while still passing as much of the incident radiation as possible. The two properties that have the greatest effect on how well the cover performs are the refractive index, which affects the losses due to reflection, and the extinction coefficient which affects how much of the radiation is



absorbed in the cover (6). These properties are approximately constant for our use even though they are functions of the wavelength of radiation (4).

For a single cover plate, the transmittance can be found using the relation (6)

$$\tau = e^{-KL} \left( \frac{1-\rho}{1+\rho} \right) \quad (3)$$

where  $L$  = the length of the path of the light through the material

$K$  = the extinction coefficient for the cover material ( $K = 0.161 \text{ cm}^{-1}$  for most glass)

$\rho$  = the reflectivity of the material

For incident radiation normal to the surface, the reflectivity can be found using the refractive index,  $n$ , by (4)

$$\rho = \left( \frac{n-1}{n+1} \right)^2 \quad (4)$$

For other angles, the following can be used

$$\rho = \frac{1}{2} \left[ \frac{\sin^2(\theta_r - \theta_i)}{\sin^2(\theta_r + \theta_i)} + \frac{\tan^2(\theta_r - \theta_i)}{\tan^2(\theta_r + \theta_i)} \right] \quad (5)$$

where

$\theta_i$  = the angle of incidence

$\theta_r$  = the angle of refraction

For most glass, the refractive index is 1.526. Since the angle of the incident radiation from the sun slowly changes throughout the day, the parameters are usually calculated for a normal angle of incidence. Correction factors for other than normal angles of



incidence can be found by using graphs in reference 6, page 32.

For a collector plate of absorptivity,  $\alpha$ , with a single cover plate, the amount of incident energy absorbed, including consideration for multiple reflection, is given by (6)

$$F_c = 1.008 \tau \alpha \quad (6)$$

If one considers the slight increase in the temperature of the cover plate due to the energy absorbed, an effective transmissivity-absorptivity product,  $F_e$ , can be found to be (6)

$$F_e = F_c + 0.23(1 - e^{-KL}) \quad (7)$$

Shading of the collector plate by the sides of the collector's enclosure, as well as any intermediate cover supports, must be taken into account. This shading factor,  $s$ , usually taken as three per cent, varies with the time of day and must be corrected accordingly (6).

Since all of these factors also vary with the angle of incidence, the calculations are normally made for a normal angle of incidence and correction factors are then applied. These are given in reference 6 on page 32.

The final factor that has an effect on how much energy reaches the collector plate is any dirt which may collect on the cover plate. This dirt factor,  $D$ , is usually assumed to be about two per cent (6).

Considering all the factors, the final amount of energy reaching the collector plate is given as (4)

$$S = HRF_e(1-D)(1-s) \quad (8)$$



The rate at which useful energy can be collected depends on the collector losses. These losses are usually calculated using a collector loss coefficient,  $U_L$ , which is the sum of the top loss coefficient,  $U_t$ , the back loss coefficient,  $U_b$ , and the edge loss coefficient,  $U_e$ .

The top loss coefficient, which has the greatest effect on losses, is due to both radiant and convective heat transfer between the collector plate and the cover plate. For a single glass cover, the heat loss is dependent upon such factors as the collector plate temperature, its emissivity, the wind speed, the ambient air temperature, and the transmissivity and emissivity of the cover plate. The top loss coefficient can be found using the relation (4)

$$U_t = \left( \frac{1}{h_{p-c} + h_{r,p-c}} + \frac{1}{h_w + h_{r,c-s}} \right)^{-1} \quad (9)$$

where

$h_{r,p-c}$  = the radiation coefficient from the collector plate to the cover plate

$h_{p-c}$  = the convective coefficient between the collector plate and the cover

$h_{r,c-s}$  = the radiation coefficient between the cover to the sky

$h_w$  = the wind convection coefficient

Methods of calculating these coefficient can be found in reference 4 on pages 129 and 130. Since the top loss coefficient and the cover plate temperature are interrelated it is necessary to estimate



a cover plate temperature and go through an iterative process to find  $U_t$ . Fortunately plots of the top loss coefficient and correction factors are available in reference 4 on pages 134 through 136.

The back loss coefficient,  $U_b$ , usually has a smaller effect on the overall losses and is normally easy to calculate. There are two main resistances to heat loss through the back of the collector, the conduction through the insulation and the convection heat transfer on the outside. The conductivity term is usually so much larger than the convective term that normally only the resistance of the insulation is used in finding  $U_b$ . Hence the back loss coefficient can be expressed as (3)

$$U_b = \frac{k_i}{L_i} \quad (10)$$

where

$k_i$  = the thermal conductivity of the insulation

$L_i$  = the thickness of the insulation

Sufficient insulation should be used to keep the back loss to about ten per cent of the top loss (6).

The edge losses are rather difficult to evaluate, but are normally small enough that a good approximation is sufficient. A reasonably close value for  $U_e$ , the edge loss coefficient, can be found using (4)

$$U_e = \frac{A_e}{A_c} U_b \quad (11)$$



where

$A_e$  = the area of the edge of the collector

$A_c$  = the frontal area of the collector

Therefore, summing all of the individual loss coefficients the final loss coefficient can be found by

$$U_L = U_t + U_b + U_e \quad (12)$$

Next, a method of determining how effectively the collector transfers the collected energy to the working fluid needs to be determined. In considering a tube on sheet plate, as shown in Figure 2, if the plate is considered to be a pair of fins attached to the tube, the amount of energy conducted to the base of the tube is (4)

$$q' = F(B-d_o)[S - U_L(T_b - T_a)] \quad (13)$$

where

$F$  = the fin efficiency

Other terms in equation 13 were defined previously. The fin efficiency is given by (4)

$$F = \frac{\tanh m(B-d_o)/2}{m(B-d_o)/2} \quad (14)$$

where

$$m = \frac{U_L}{k\delta} \quad (15)$$

If the energy collected directly by the tube is considered, the total useful energy gain per unit of length of collector is (6)

$$q'_u = [(B-d_o)F + d_o][S - U_L(T_b - T_a)] \quad (16)$$



Next, the bond conductance,  $C_b$ , should be considered since it has a large effect on how much of the energy collected by the plate gets to the tube. The bond conductance is bound by

$$C_b = \frac{k'b}{\gamma} \quad (17)$$

where

$k'$  = the bond thermal conductivity

$b$  = the bond length

$\gamma$  = the average bond thickness

An alternate method for calculating the useful energy gain in terms of known or measurable parameters can be expressed as

$$q' = F'[S - U_L(T_f - T_a)] \quad (18)$$

where

$F'$  = the collector efficiency factor

The collector efficiency factor,  $F'$ , is defined as the ratio of an overall heat transfer coefficient between the heat removal fluid of the collector and the ambient air,  $U_o$ , and the collector loss coefficient,  $U_L$ , that is

$$F' = \frac{U_o}{U_L} \quad (19)$$

For collectors with tubes attached above the collector plate which is the method used to construct the collector described in Chapter III.,  $F'$  is given by

$$F' = \left[ \frac{BU_L}{\pi dh} + \frac{1}{\frac{d_o}{B} + \frac{1}{\frac{BU_L}{C_b} + \frac{B}{2L'F}}} \right]^{-1} \quad (20)$$



The last factor which influences the rate at which useful energy can be collected is the flow factor,  $F''$ , which can be found using (6)

$$F'' = \frac{Gc_p}{U_L F'} \left[ 1 - \exp\left(\frac{U_L F'}{Gc_p}\right) \right] \quad (21)$$

where

$c_p$  = the specific heat of collection fluid

$G$  = the fluid flow rate per unit of collector area

In summary a generalized performance equation for the useful energy collection rate of the collector may be written as

$$q_u = F' F'' [S - U_L (T_i - T_a)] \quad (22)$$

where

$F''$  = the flow factor

$F'$  = the collector efficiency factor

$S$  = the amount of energy reaching the collector plate

$U_L$  = the overall loss coefficient

$T_i$  = the inlet fluid temperature

$T_a$  = the ambient air temperature

As a measure of a collectors performance, its efficiency,  $\eta$ , can be determined as the ratio of useful energy gain to the total incident energy (4), that is

$$\eta = \frac{\int \frac{Q_u}{A_c} dt}{\int HR dt} \quad (23)$$



where

$A_c$  = the frontal area of the collector

$Q_u$  = the total useful energy gain

$H$  = the rate of incident radiation

$R$  = factor to convert beam or diffuse radiation  
to that on the plane of the collector

$dt$  = the time period of interest

Since this would be difficult to use the efficiency is normally calculated as shown in Chapter IV.

In conclusion, a set of equations has been presented that allows the calculation of the theoretical useful energy gain of a collector and its efficiency. The equations given also help to understand the important interactions of the various factors involved in the collector design.



## CHAPTER II

### ANALYTICAL MODEL OF A FLAT PLATE

#### HEAT PIPE SOLAR COLLECTOR

The use of heat pipes to transport heat energy from the absorber plate of a solar collector to the air in a space heating system offers several advantages, as stated earlier in Chapter I of this thesis. The two most important advantages are the protection from freezing and the diode action of a wickless heat pipe. To evaluate these advantages, a heat pipe solar collector was designed, fabricated and tested.

There are several ways of combining a flat plate collector and heat pipes. The most obvious was is to attach one or more heat pipes to an absorber plate. A second method is to use an extrusion in which the heat pipe is an integral part of the collector plate. A third method, which is similar to the second, is an absorber made of two plates that are bonded together to form a single plate with internal passages formed. This is called a Roll Bond panel, which is a registered trademark. The passages of a Roll Bond panel can be used as the heat pipe passages. As Figure 3 shows, the second and third methods have the advantage of eliminating the bond conductance between the plate and the pipe. However, from an economic



viewpoint extrusions do not appear to be cost effective when compared against the other two methods.

With any of these methods, an efficient method for transferring the collected energy to the working fluid of a distribution system must be considered. In the extruded collector, there is a need for some sort of condenser-heat exchanger to be attached. Also, each heat pipe would have to be charged individually. Although the extrusions would eliminate the problem of the bond conductance the increase in cost associated with their fabrication does not appear justifiable (7).

The Roll Bond panel also has the advantage of eliminating the bond conductance resistance. In addition, as Figure 4 shows, the panel could be evacuated and charged through the common manifold. This type of collector would also require the attachment of a separate heat exchanger.

Individual heat pipes fastened to a collector panel have several advantages over the above methods. The design of the collector is much more straight forward since the pipes are clamped, soldered or brazed to the plate. The advantage of this is that commercially available precharged heat pipes can be utilized. The heat pipes can be purchased with heat transfer fins already attached to the condenser section thus eliminating the problem of attaching a heat exchanger to the heat pipes. However, this may result in the disadvantage of an added thermal resistance of the bond between the plate and the heat pipe. This thermal resistance can be negligible



if the collector plate is formed to partially wrap around the pipe and carefully soldered to the pipe (7). Care must be exercised to insure that a good thermal bond is obtained during the soldering or brazing process. In general the bond resistance should be less than  $\frac{0.02\text{BTU}}{\text{hr-ft}^2\text{F}}$  (4). After consideration of the advantages and disadvantages of the different methods, the soldering of individual heat pipes to a copper collector panel was chosen for the construction of the 2' x 5' prototype heat pipe flat plate collector. The condenser section of the heat pipes were fitted with fins in order to transfer the collected energy to the working fluid used in the distribution system.

Having selected the type of collector, equations describing the performance of a heat pipe collector can be determined. The main difference that must be taken into account is the added thermal resistance of transferring the heat from the heat pipe to the working fluid in the condenser manifold system. Equation 20 can be modified to take into account the additional thermal resistance between the condenser section and the working fluid in the air duct. This modification results in (7)

$$F' = \left[ \frac{BU_L}{\pi dh} + \frac{1}{\frac{d_o}{B} + \frac{1}{\frac{BU_L}{C_b} + \frac{B}{2L'F}}} + U_L HBR_h \right]^{-1} \quad (24)$$

where

$h$  = the evaporation film coefficient

$B$  = the center to center distance between



heat pipes

$H$  = the total height of the panel

$R_h$  = the resistance of the interface between a  
heat pipe and the air

The other terms of Equation 24 were defined previously in Chapter II of this thesis.  $R_h$  can be determined by the following

$$R_h = \frac{1}{A_f h_f} \quad (25)$$

where

$A_f$  = the total area of the heat transfer fins

$h_f$  = the convective heat transfer coefficient for  
the fins to the air

The term  $h_f$  will vary with the geometry and the operating conditions. A handbook such as Perry's Chemical Engineers Handbook should be consulted to determine the value of  $h_f$  for a particular situation. For our situation, a row of finned condensers,  $h_f$  can be found using (8)

$$h_f = 0.17 \frac{(V_F)^{0.6}}{(d_o)^{0.4}} \left( \frac{B}{B-d_o} \right)^{0.6} \quad (26)$$

where

$V_F$  = the face velocity of the air, in ft/min

$d_o$  = the outside diameter of the bare pipe, in  
inches

It should be noted that the bond conductances between the absorber plate and the heat pipe, and the heat pipe and the heat transfer



fins, as well as the convective heat transfer from the fins to the air, can all have a detrimental affect on the collector's efficiency.

The flow factor, for a heat pipe collector, is the same as for a conventional collector and can be found using Equation 21. The mass flow rate used would be that of the air. For a heat pipe collector with an air distribution system which totally encloses the condenser fin section, the recommended flow rate per unit of collector area is 2 CFM/ft<sup>2</sup> (9).

The generalized performance equation can now be written using the factors just given, the overall loss coefficient, and certain measurable quantities. This was given by Equation 22 and is repeated here, as follows

$$q_u = F'F''[S - U_L(T_i - T_a)] \quad (26)$$

With this equation and knowing the operating conditions, it is possible to calculate the theoretical useful energy gain.

The only limit imposed by the use of the heat pipes is their capacity to transport the collected energy. The heat pipes chosen are rated by the manufacturer to be able to transport 300 watts. Since the maximum amount of energy, as shown in Chapter I, that is available for collection by an individual heat pipe section is much less than this, it should not be limiting factor.



## CHAPTER III

### CONSTRUCTION

The heat pipe collector was constructed using four individual heat pipes each six feet long. The pipes were made of 5/8 inch hard drawn copper tubing that were charged with water, as the working fluid, after having had their internal pressure reduced to less than 300 microns. Each heat pipe was soldered to a strip of copper, six inches wide and sixty inches long. Each copper strip had a valley formed down its center as shown in Figure 8. The purpose of their groove was to reduce the bond conductance as much as possible. The first six inches of each heat pipe acted as the condenser section. These first six inches of the heat pipe had a spiralling fin soldered around it as shown in Figure 8 to transfer the heat energy from the heat pipe to the air. The next six inches were the adiabatic section. The remaining five feet of the heat pipe formed the evaporator section. the heat pipe-collector plate combinations were held in place by clamping them between wood strips. The wood strips were then fastened to the aluminum enclosure.

The collector box was constructed of a sheet of 1/16 inch aluminum formed into a box 29 inches wide, 64.5 inches long and



2.5 inches deep. The box was insulated with one inch thick Technifoam 400, a rigid foam insulation faced with aluminum foil. The cover plate was  $3/16$  inch double strength plate glass. Foam weather stripping was applied to seal the cover plate to the collector box.

Heat was removed from the heat pipes by air passing over the fins of the heat pipes. The fins were enclosed in a duct constructed of Technifoam. The air was blown through the duct by a small cooling fan. The fan's speed was controlled by varying its supply voltage so that the required flow was established. For the 10 square foot collector used in this project 20 CFM was required. A converging nozzle was placed at the end of the duct to accelerate the flow to a speed that could then be easily measured. This was necessary so that a check could be kept on the flow rate. Figure 9 shows the collector before installation of the duct and Figure 10 shows the collector after installation.

The entire collector was tilted 29 degrees up from horizontal so that the incident beam radiation was nearly normal to the collector. The collector was instrumented with a six temperature sensing silicon diodes as shown in Figure 11. Details of the instrumentation system and the temperature sensing diodes are discussed in Chapter V.



## CHAPTER IV

## DESIGN OF DATA ACQUISITION SYSTEM

To evaluate any solar collector it is necessary to have accurate information on the solar insolation that the collector is exposed to and the temperatures at various points on the collector. With this data, an understanding of the collectors performance can be gained. Therefore, a method of recording temperatures at several different locations on the system was required. The method of recording the insolation of the collector was an Epply Pyranometer Model PSP which recorded both the instantaneous intensity of the solar radiation and a continuously intergrated value.

A reliable temperature sensing system was necessary for accurate evaluation of the performance of the heat pipe solar energy collector. The objective was to find a temperature sensing device that would represent temperature as a voltage signal. The voltage signals could then be recorded for later use.

One of the temperature sensing methods that was considered, but eliminated, is the widely used thermocouple. Thermocouples are, under proper conditions, consistent and reliable. The basis of operation is the Seebeck effect (10). The Seebeck effect is the generation of a voltage by maintaining two dissimilar metal junc-



tions at different temperatures. Good accuracy can be obtained due to the variation of junction voltage as it varies linearly with temperature. The major disadvantage of the thermocouple is that the voltage is very small, a maximum of 60 microvolts per degree Celsius, thus making the thermocouple very susceptible to picking up error voltages from the environment. Distorted readings may result if the thermocouple circuit receives radio signals from a nearby transmitter. An additional problem, when using a thermocouple is that high quality, carefully calibrated equipment must be used due to the size of the signal. This introduces the problem of maintaining calibration at increased costs. Also, care must be taken when working with a thermocouple to select materials which are compatible to be introduced into the measurement system. Since the system operates on the idea of dissimilar materials generating a voltage, error signals can be introduced by such types of materials. The last disadvantage is that a reference junction is necessary for the utilization of a thermocouple as a temperature sensing device, thus adding to the complexity and the cost of the system. Therefore, the use of thermocouples were rejected because of these principal reasons, as well as the cost and difficulty of using them on a solar collector which would tend to act as an antenna for error signals.

Resistance thermometers are another common means of making temperature measurements. Their principle of operation is based upon the linear variation of electrical resistance with temperature.



This phenomenon normally maintains a linear variation over a wide temperature range. Like the thermocouple it will, under the right conditions, provide good reliable measurements. However, this method necessitates either special equipment for reading out the temperatures or a bridge circuit to determine the resistance. This is a major disadvantage since this type of special equipment is expensive and the bridge circuit requires consulting a calibration curve. Since the lead wire resistance varies with temperature one must keep the lead wires short in order to avoid erroneous measurements (10). The resistance thermometer was ruled out due to the problems of general inconvenience and special handling to avoid erroneous measurements.

Another sensor that uses the idea of resistance varying with temperature is the Thermistor. The Thermistor is generally used in temperature compensating circuits. Unlike the conductor type sensors, this semi-conductor's resistance is inversely related to its temperature. The major disadvantage of the Thermistor is its nonlinearity. This nonlinear nature of the Thermistor makes it difficult to use with data recording equipment. Another disadvantage is that care must be taken when purchasing a Thermistor that it has been properly aged, since the device's response will vary with age (10). Hence, the Thermistor was rejected due to nonlinearities and its deterioration with age.

A temperature sensitive semi-conductor that has the needed linear variation with temperature is the common silicon diode.



The principle of operation of the diode is the variation of the reverse saturation current with temperature. The main advantage of a silicon diode is the size of its output voltage and the significant change with the temperature which it represents. As a result, the system is resistant to error signals and can have a long lead wire to the sensor probe. In addition, due to the large output, less sensitive and hence less expensive equipment can be used to record the data. Finally, the circuit is simple to design and the components are inexpensive.

The basis for practical operation of the diode system is that the diode voltage is related to the reverse saturation current which varies with temperature. For the temperature range of interest in solar collector applications,  $-50^{\circ}$  to  $+175^{\circ}$  Celsius, the variation is linear and the sensitivity is comparable to a thermocouple. The relation for a silicon diode is given by (11).

$$V_D = \frac{NkT}{q} \ln\left(\frac{I_D}{I_r(T)} - 1\right) \quad \text{volts} \quad (27)$$

where

- $V_D$  = diode voltage
- $I_D$  = diode current
- $k$  = Boltzman's constant
- $T$  = temperature,  $^{\circ}\text{K}$
- $q$  = electron charge in coulombs
- $N$  = geometrical constant of the junction  
( $1 \leq N \leq 2$  for silicon)



$$I_r(T) = \text{reverse current, } I_{ro} 2^{\left(\frac{T-T_0}{2}\right)}$$

$$I_{ro} = I_r(T_0)$$

This relation indicates that the diode voltage will decrease with increasing temperature. This is due to the reverse current approximately doubling for each 10 degrees Celsius rise. For a diode with the characteristics shown in Table 2, we find that the variation of voltage for a change in temperature is  $-2.97 \text{ mv}/^\circ\text{C}$ . This compares to only  $60 \text{ } \mu\text{v}/^\circ\text{C}$  for a thermocouple.

If one examines Equation 27 there are two unknowns,  $N$  and  $I_{ro}$ , that must be found in order to use a diode as a thermometer. A technique for determining these constants is discussed in detail in the following sentences. Since there are two unknowns, the simultaneous solution of two equations was required to determine their solutions. Values for  $25^\circ$  Celsius and another operating point were substituted into Equation 27 to generate the two needed equations. These equations are

$$V_{D1} = N \frac{kT}{q} \ln[(I_{D1}) - \ln(I_{ro})] \quad (28)$$

$$V_{D2} = N \frac{kT}{q} \ln[(I_{D2}) - \ln(I_{ro})] \quad (29)$$

Subtracting Equation 29 from Equation 28 yields

$$V_{D1} - V_{D2} = N \frac{kT}{q} \ln\left[\frac{I_{D1}}{I_{D2}}\right] \quad (30)$$

which gives upon solving for  $N$ ,

$$N = \frac{q(V_{D1}-V_{D2})}{kT \ln(I_{D1}/I_{D2})} \quad (31)$$



Using Equation 31 in Equation 28 or 29 gives the following result

$$I_r (25^{\circ}\text{C}) = I_D \exp \left( \frac{-V_{DQ}}{NkT} \right) \quad (32)$$

Using Equation 32, a calibration curve can be constructed by supplying the diode with a constant current and varying its temperature over the desired operating range. A typical calibration curve is shown in Figure 5.

In order to make the voltage signal easier to work with, a preamplifier was used. Such a preamplifier should have an offset voltage of less than two millivolts in order to maintain one degree Celsius accuracy. A low cost operational amplifier, such as an LM741CN, can deliver a gain of 11 and usually has a one millivolt offset. Figure 6 shows a practical circuit, including resistor R1 which acts as the current source.

In the actual system designed, it was necessary to record temperature readings from several points in the solar collector. Therefore, it was desirable if the collector temperatures, as well as the ambient temperature, could be recorded in a single device. In order to accomplish this task a Complimentary Metal-Oxide Semiconductor, CMOS, transmission gate was used to multiplex the various signals on to a single line. One simple and inexpensive CMOS unit is a 4051PC, which contains the circuitry to decode an instruction signal and choose any of eight channels and place it on an output line. Two of these units can be used together to obtain sixteen multiplexed channels. Two of these channels can be used for reference signals and still have the ability to take fourteen different



readings. A timing circuit can generate the addressing signal needed to control the multiplexers and set the sample period. The timers used consisted of an RC circuit and one or two CD4020AE fourteen bit counters. A second timer was used to set the sample period. As an added benefit, this second timer also controlled the recording device. A strip chart recorder was chosen as the recording device. The second timer was used to control the recorder in order to prevent excessive waste of chart paper. A recording speed of four inches per hour over an eight hour run resulted in only thirty two inches of paper being used. In the design shown in Figure 7, untrimmed quad amplifiers (SN721044) were used. These units had the benefits of an offset voltage small enough that it need not be trimmed and that each unit conveniently contained four amplifiers. These fed into the multiplexers and on into a second amplifier, a LM741CN, which is capable of delivering ten milliamperes to a 1000 ohm resistance.

Even though the silicon diode is relatively consistent, there is an offset voltage factor that must be considered. Matching the offset voltages of the diodes eliminated this problem. The offset voltages were matched by measuring the voltage response of a large number of diodes at a constant temperature. The seven matched diodes that were needed for the system were found by grouping the diodes into their similar voltage response categories.

Before being put into operation, the data system had to be checked out. Temperature measurements were taken and compared to



measurements taken with an accurate mercury thermometer. In checking the system out shorts and misconnections in the circuits were the main problems. An oscilloscope and a volt meter were used to locate the problems. Once the circuitry problems were found they were easily corrected by rewiring. The system was used for two months of nearly continuous use and proved completely reliable with consistent  $1^{\circ}\text{C}$  accuracy. In spite of exposure to a nearby radio transmitter, there were no indication of erroneous readings even with the use of long lead wires.

While the system proved very satisfactory, there are improvements that would enhance the accuracy of the unit and make it more convenient to use. Greater accuracy could be obtained by supplying the diode with a more stable current supply by use of a feedback circuit. Also, a probe of three diodes in series would result in greater accuracy. Table 3 shows an illustration of the accuracy that can be obtained by this method. Accuracy could be improved and the need to match the diodes eliminated by trimming the preamplifiers offset voltage to zero. The offset voltage can be trimmed to zero by using a variable resistor as shown in Figure 6. There could be improvement in the quality of the preamplifiers by use of SN721084 preamplifiers. These are higher performance amplifiers than the ones currently being used in the prototype data recording system. Since the time of the temperature reading is important, a more precise timer would be beneficial. Replacing the resistor-capacitor pair with a crystal frequency generator would



greatly improve the timing of the device. Finally, a means of converting the analog signal to a digital one and recording it would be helpful. This would make the data available for immediate use and could be put into a form for direct input into a computer for data reduction.

The application of silicon diodes as temperature sensing devices for solar energy collectors proved to be reliable, accurate, and economical. The availability of fourteen channels plus the ability to control the recorder proved to be a viable solution to the problem. The cost of the entire system was less than fifty dollars. The many advantages over other temperature sensing devices showed the silicon diode to be superior for use with the solar collector. The simple design and theory of the silicon diode has few negative aspects and furthermore, with certain improvements the system has the potential for even greater accuracy. Figure 12 shows the entire data system as used.



## CHAPTER 5

## THERMAL PERFORMANCE OF THE HEAT PIPE COLLECTOR

In order to evaluate the thermal performance of the collector it was necessary to record information about the incident solar radiation, the ambient air temperature, and the temperature of several points in the collector system. The actual data system, which was discussed previously, was connected to the collector as shown in Figure 11.

The ambient air temperature was measured by probe number 1. The other temperature sensors were located as shown in Figure 11. The drop in temperature across the collector plate to heat pipe bond was found using sensors 2 and 3. The drop in temperature due to the adiabatic section of the heat pipe was found by taking the difference in the temperature measurements of sensor 3 and sensor 4. The temperatures measured by sensor 4 and sensor 5 gave an indication of the condenser's performance. Sensor 6 was used to measure the air temperature before passing through the row of condensers and sensor 7 measured the temperature after the condensers. Using this information and knowing the flow rate of 20 CFM through the collector the actual useful energy gain of the collector, given by Equation 26, was calculated using the numerical



approximation

$$Q_u = \dot{m} c_p \overline{\Delta T} \Delta \tau \quad (33)$$

where

$\dot{m}$  = the mass flow rate of the air

$c_p$  = the specific heat of air

$\Delta \tau$  = the time period of interest

$\overline{\Delta T}$  = the average difference between the outlet  
air temperature and the ambient air temperature

Thirty minute time periods were used and the specific heat of the air was assumed to be constant over the temperature range measured.

This allowed Equation 33 to be reduced to

$$Q_u = 18.4 (\overline{\Delta T}) \text{ BTU} \quad (34)$$

The theoretical useful energy gain was calculated using Equation 26 with the value for  $S$  determined by (6)

$$S = 0.93 (F_{e\theta=0}) \left[ \frac{F_{e\theta}}{F_{e\theta=0}} \right] \left[ \frac{(1-s)\theta}{(1-s)\theta=0} \right] I \quad (35)$$

where

$F_{e\theta}$  = the effective transmittance-absorptance  
product for the angle of incidence  $\theta$

$\theta$  = a subscript describing the angle of incidence for beam radiation,  $\theta=0$  is a normal angle of incidence

$I$  = the incident solar radiation measured by the  
Pyranometer



The two correction factors are obtained from graphs in Reference 6 on page 32. A method for determining the angle of incidence is outlined in Reference 4.

Knowing the energy gain and the total incident radiation striking the collector,  $Q_s$ , the efficiency of the collector,  $\eta$ , can be calculated. Equation 23, which defines the efficiency, is used in the modified form

$$\eta = \frac{Q_u}{Q_s} \quad (36)$$

where

$Q_u$  = either the actual or theoretical useful energy gain depending upon whether the actual or theoretical efficiency is of interest

$$Q_s = I A_c \Delta \tau$$

The actual value for  $Q_s$  was found using the integrated value recorded by the Pyranometer by using the relation

$$Q_s = (10 \text{ ft}^2) (7.354 \frac{\text{BTU}}{\text{ft}^2 \text{ penstroke}}) \left( \frac{\text{no of penstrokes}}{\text{for the time period } \Delta \tau} \right) \quad (37)$$

Values for both the actual useful energy gain and the actual efficiency were calculated for each day's data. Also for May 8, 1977, the theoretical useful energy gain was calculated. Figure 13 shows the plot of the actual and theoretical useful energy gain versus time.

The collector was mounted facing south on the roof of the Engineering building at Florida Technological University and was



free of shadows for the middle 8 hours of each day. To give the best performance the collector was tilted up to an angle equal to the locations latitude ( $28\frac{1}{2}^{\circ}$ ).

When evaluating the experimental data it was necessary to check for an indication to determine if the fan in the duct system had come to a complete stop. This was indicated by the temperature of sensor 6 being higher than sensor 7. This normally was a result of a gust of wind striking the collector's duct system obliquely at the duct exit. This resulted in a back flow of ambient air across the fan thereby bringing it to a complete stop. Several data points and in some cases a total day's run had to be discarded because of this phenomenon. To avoid this problem the data presented was collected on relatively calm days where the wind speed was steady and less than approximately 5 meters per second. The data for a cloudless day is shown in Figure 14 and Tables 5 and 6. The data for an average day is shown in Figure 15 and Table 7. Figure 16 and Table 8 show the data for a cloudy, overcast day with intermittent rain.



## CHAPTER VI

## CONCLUSIONS AND RECOMMENDATIONS

The data presented shows that the actual amount of useful energy transferred to the air flow and the average efficiency of the collector are lower than expected. There appears to be three factors which caused these low values. These factors are:

1. poor bond conductance from the absorber plate to the heat pipe.  
This may be caused by undetected voids in the solder joints.  
If the plate's shape conformed to the shape of the pipe more closely, a thinner solder joint could be used to reduce the bond resistance.
2. there is approximately a five per cent loss in the energy transferred from the evaporator to the condenser (12). There is also energy loss by heat transfer from the heat pipe to the collector's housing by conduction through the wood used to hold the pipes in place. This is indicated by the difference in the heat pipe's temperature just before the wood and at the beginning of the condenser which is less than six inches away. This temperature difference was as large as  $9^{\circ}\text{C}$ .
3. poor heat transfer to the air by the fins, possibly caused by poor air flow over the fins on the downstream side of the heat



pipes. The fact that the temperature upstream of the heat pipes is higher than the ambient air temperature indicates that the air flow is not straight through the heat exchangers in the duct and out the duct as expected.

As a project nears completion, there are always ideas for improvements that develop. These ideas usually result from having a better understanding of the problems involved. In the heat pipe collector, the elimination of the bond conductance for the heat pipe to collector plate joint would greatly improve the collectors performance. Although the thermal resistance was thought to be negligible during the design of the collector the resistance commonly caused a  $15^{\circ}\text{C}$  drop in temperature. A solution to the problem would be to use a bonded collector plate. Such panels are available from the Olin Brass Company under the trade name of Roll-Bond Panels.

Another improvement would be to design the collector's heat pipes to use gravity to return the condensate to the collector-evaporator section, rather than using a wick. If this is done, the heat pipes will act as thermal diodes thus improving the amount of energy collected during a partly cloudy day when the insolation is intermittent.

Since the use of a bonded collector plate would necessitate attaching a condenser-heat exchanger, the heat exchanger can be designed for optimum performance. This would include increasing the area of the condenser and of the fins used to transfer the heat



to the air. The design of the heat exchanger could also provide for the fins to receive maximum exposure to the air stream. A related improvement would be the use of thin tubes to connect the heat pipes of the collector and their condensers. This would be done to minimize the system losses due to conduction of heat from the condenser back to the collector. This would be beneficial during a cloudy period when the absorber plate might be cooler than the air in the distribution system.

The heat transfer from the condenser could also be improved by a duct design that forces the air to flow across the heat exchangers fins. The duct should be built to insure that a uniform air flow is maintained throughout the flow.

The cost of the collector could be reduced by the use of bonded aluminum panels rather than the copper panels. The lower conductivity of aluminum will have a small effect on the performance because of the close spacing of passages in a bonded panel.

There improvements are obvious at the present time and they would do a great deal to overcome the low efficiencies of the present design.



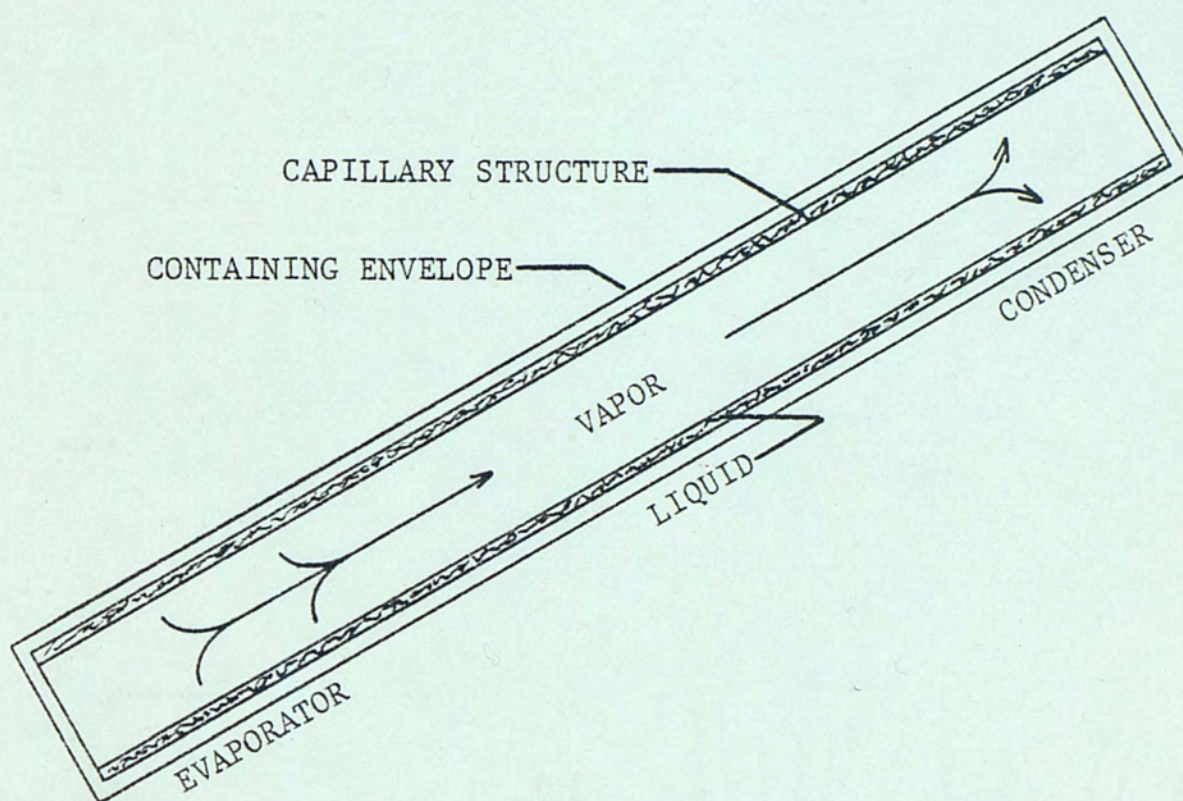


FIGURE 1. TYPICAL HEAT PIPE



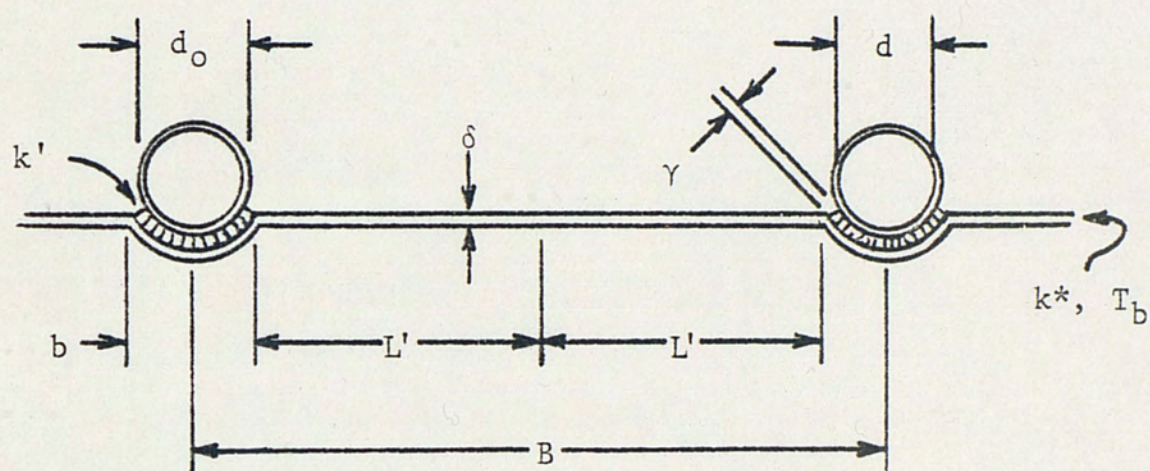


FIGURE 2. TUBE ON COLLECTOR PLATE

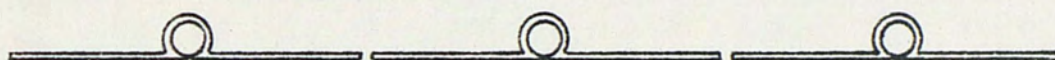


EITHER SIDE COULD BE ABSORBER SURFACE

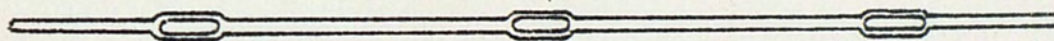


- a. INDIVIDUAL HEAT PIPES BONDED TO ABSORBER PLATE IN PREFORMED GROOVES

EITHER SIDE COULD BE ABSORBER SURFACE



- b. INDIVIDUAL HEAT PIPES EXTRUDED WITH INTEGRAL ABSORBER FINS

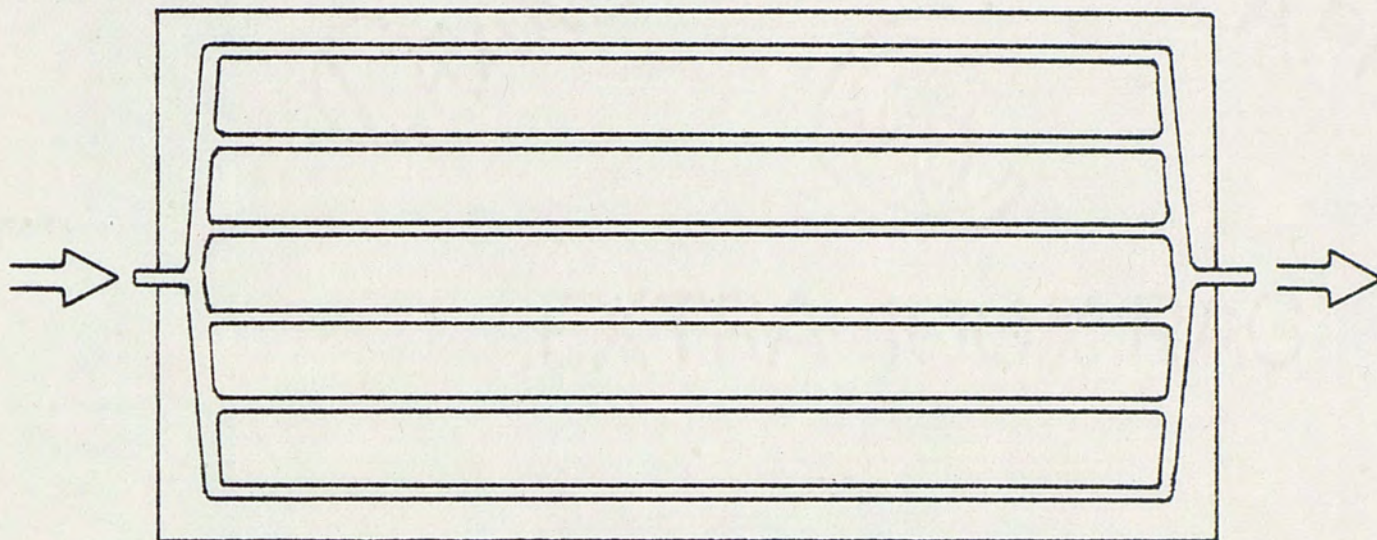


- c. ABSORBER PLATE WITH INTEGRAL HEAT PIPES FORMED IN 2-PIECE BONDED PLATE

FIGURE 3. VARIOUS HEAT PIPE COLLECTOR CONFIGURATIONS



BEFORE MODIFICATION



AFTER MODIFICATION

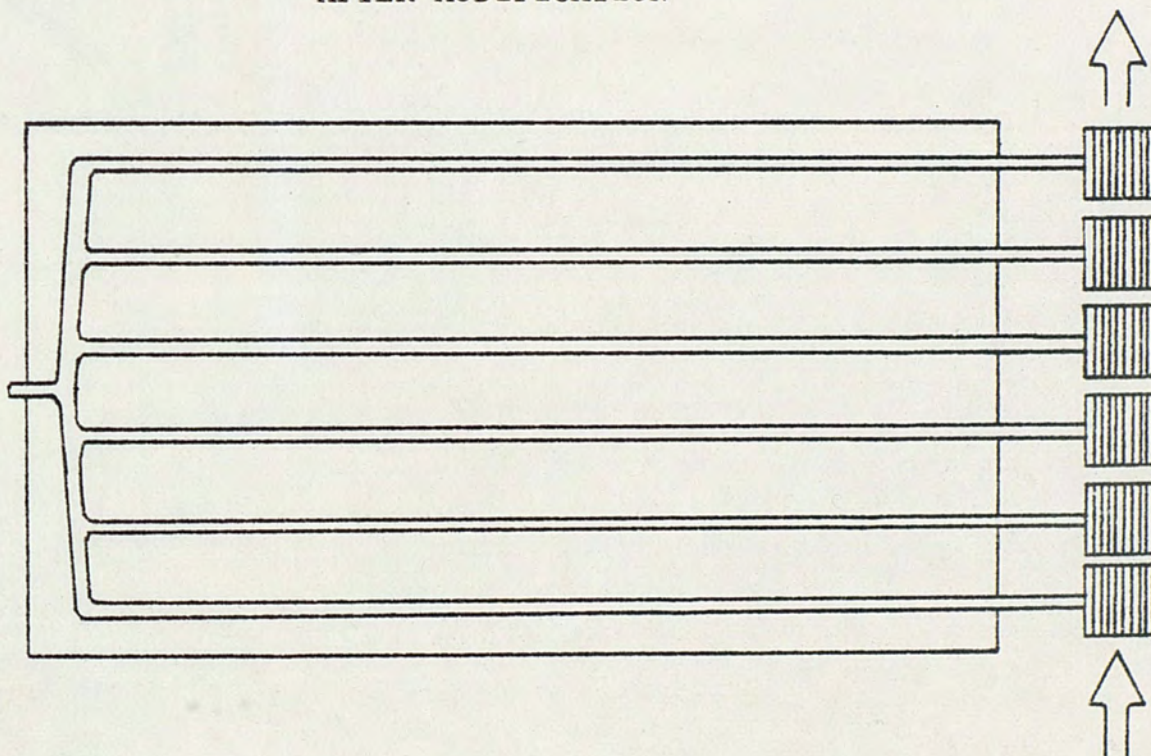


FIGURE 4. ROLL BOND PANEL BEFORE AND AFTER  
MODIFICATION FOR HEAT PIPES



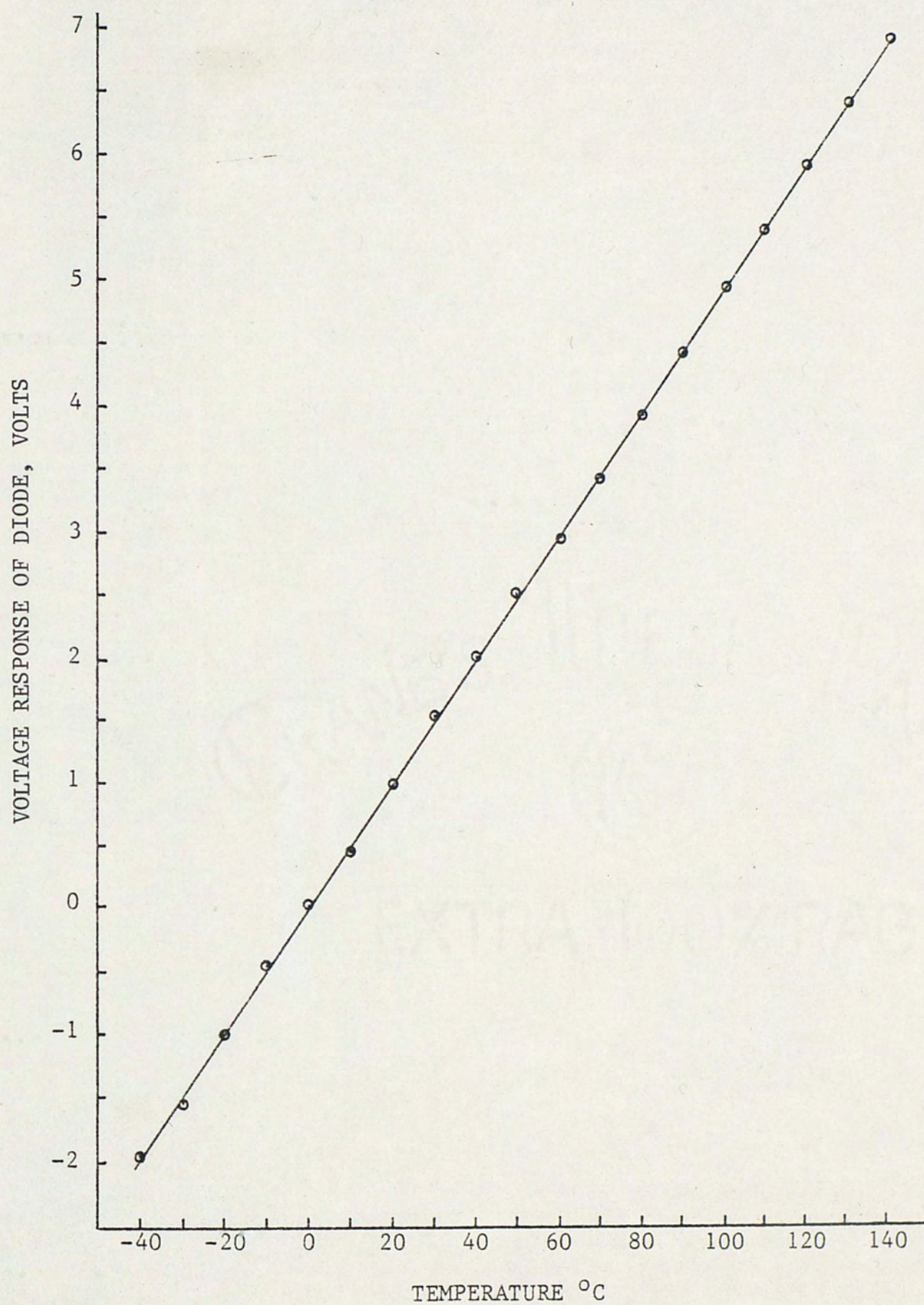


FIGURE 5. CALIBRATION CURVE FOR DIODE TEMPERATURE SENSOR



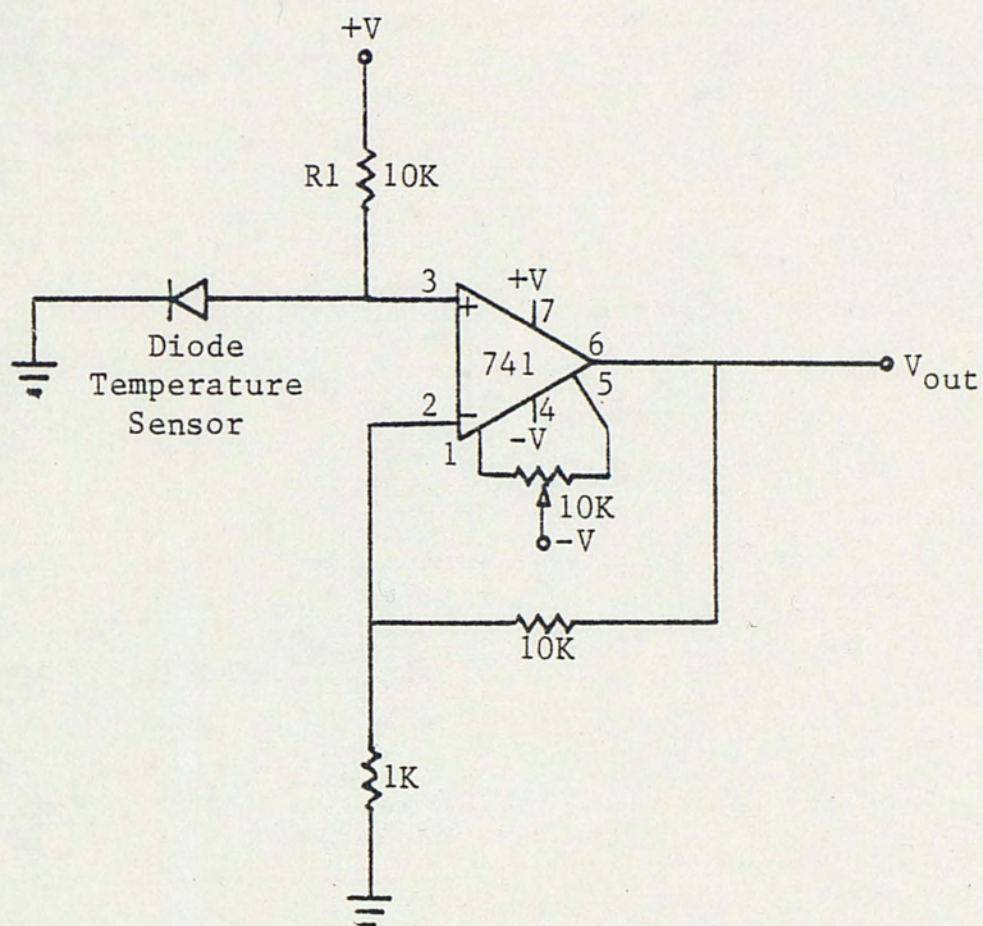


FIGURE 6. SINGLE DIODE CIRCUIT



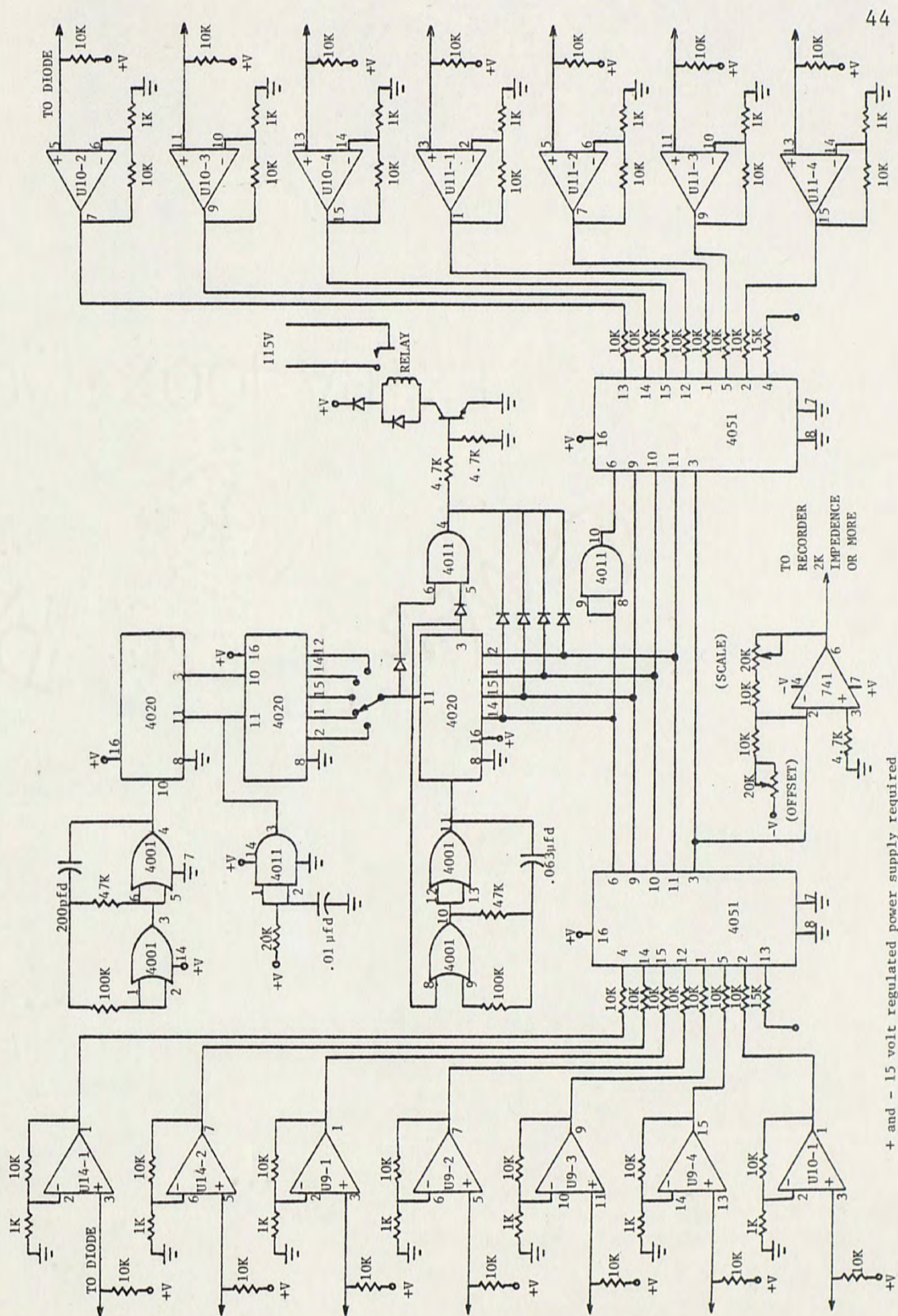
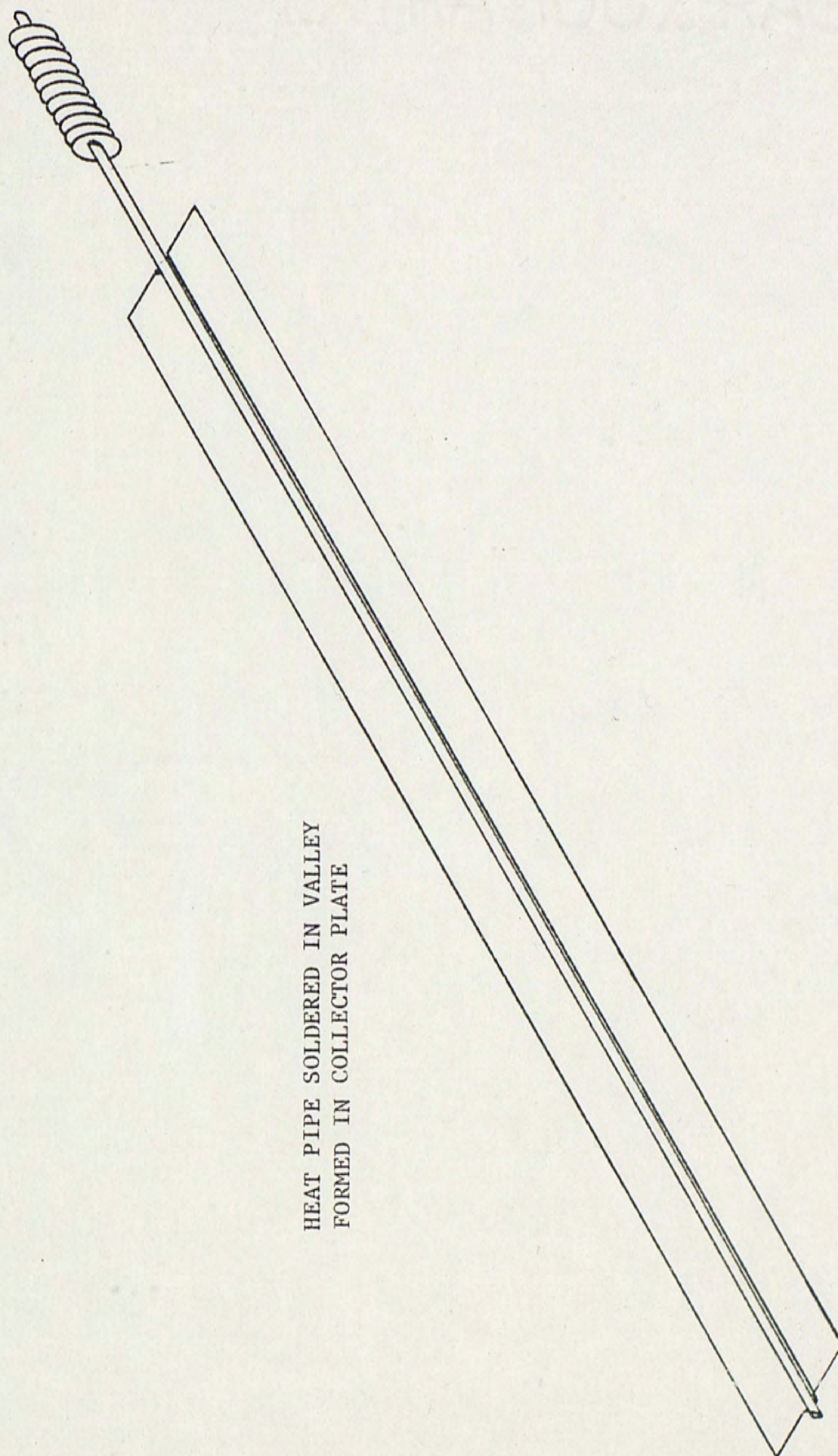


FIGURE 7. MULTIPLEXED TEMPERATURE MEASURING CIRCUIT





HEAT PIPE SOLDERED IN VALLEY  
FORMED IN COLLECTOR PLATE

FIGURE 8. HEAT PIPE ON COLLECTOR PLATE



FIGURE 9. PHOTO OF HEAT PIPE COLLECTOR WITHOUT DUCT



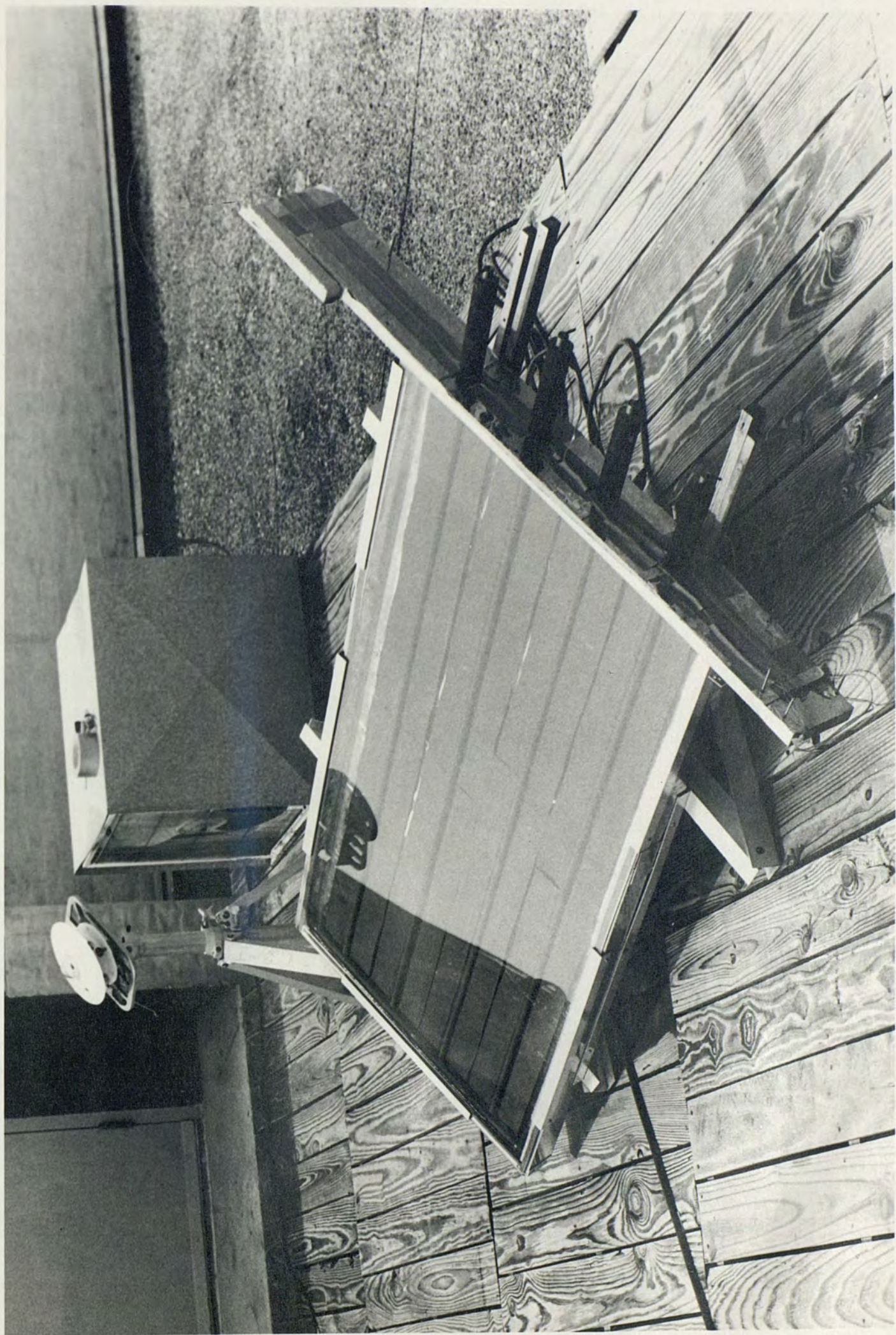




FIGURE 10. PHOTO OF HEAT PIPE COLLECTOR WITH DUCT INSTALLED







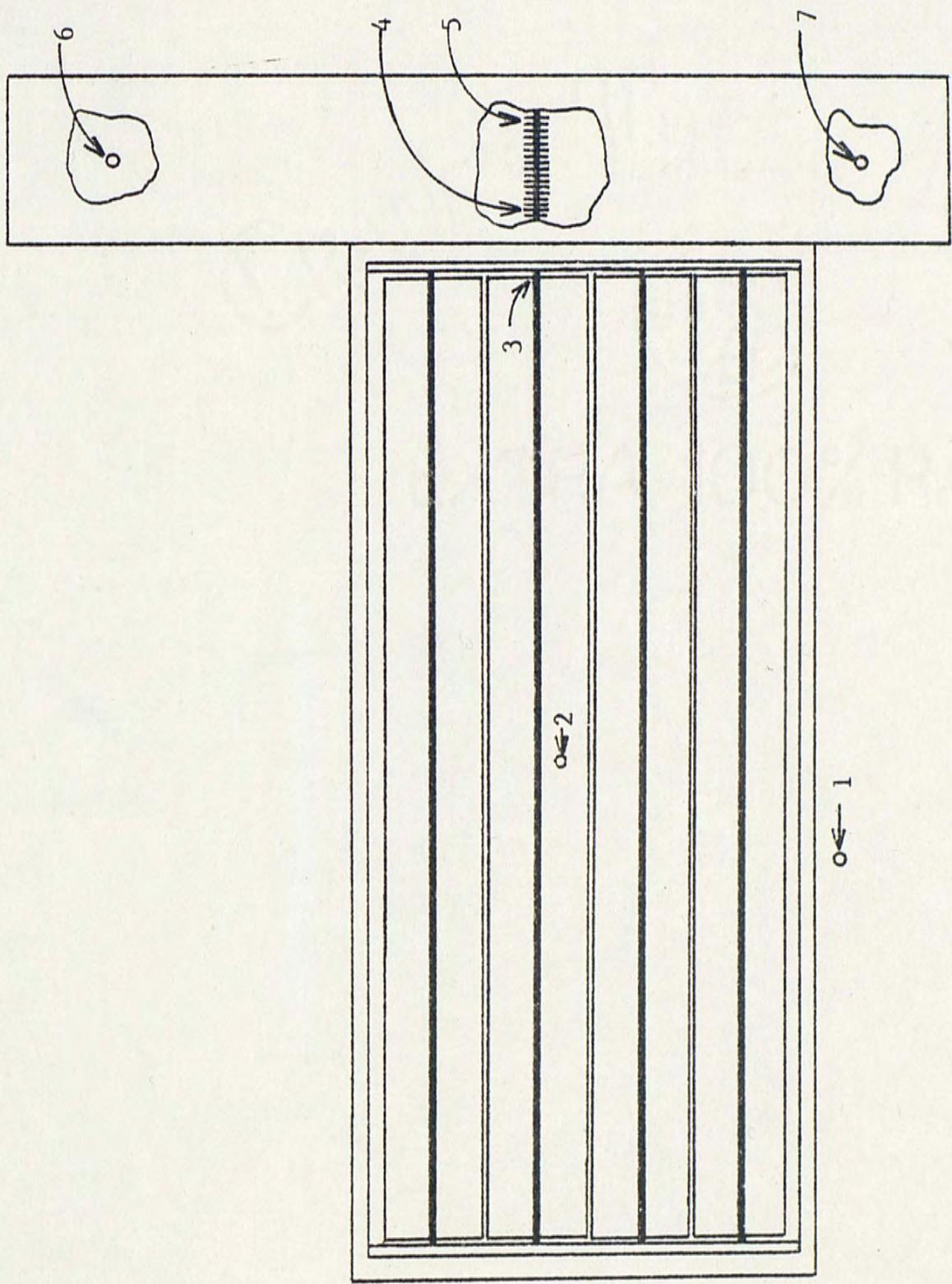


FIGURE 11. PLACEMENT OF TEMPERATURE SENSORS



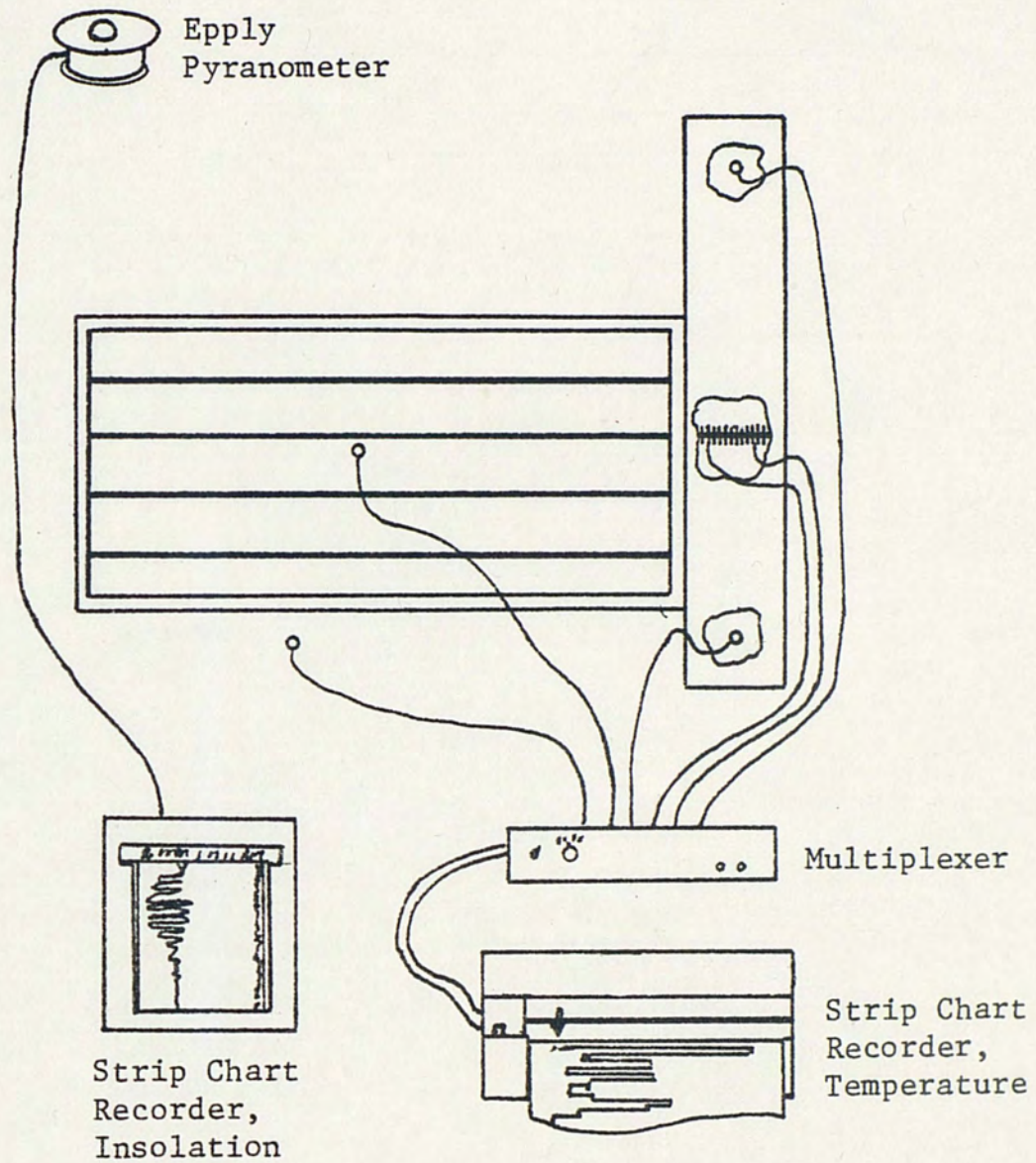
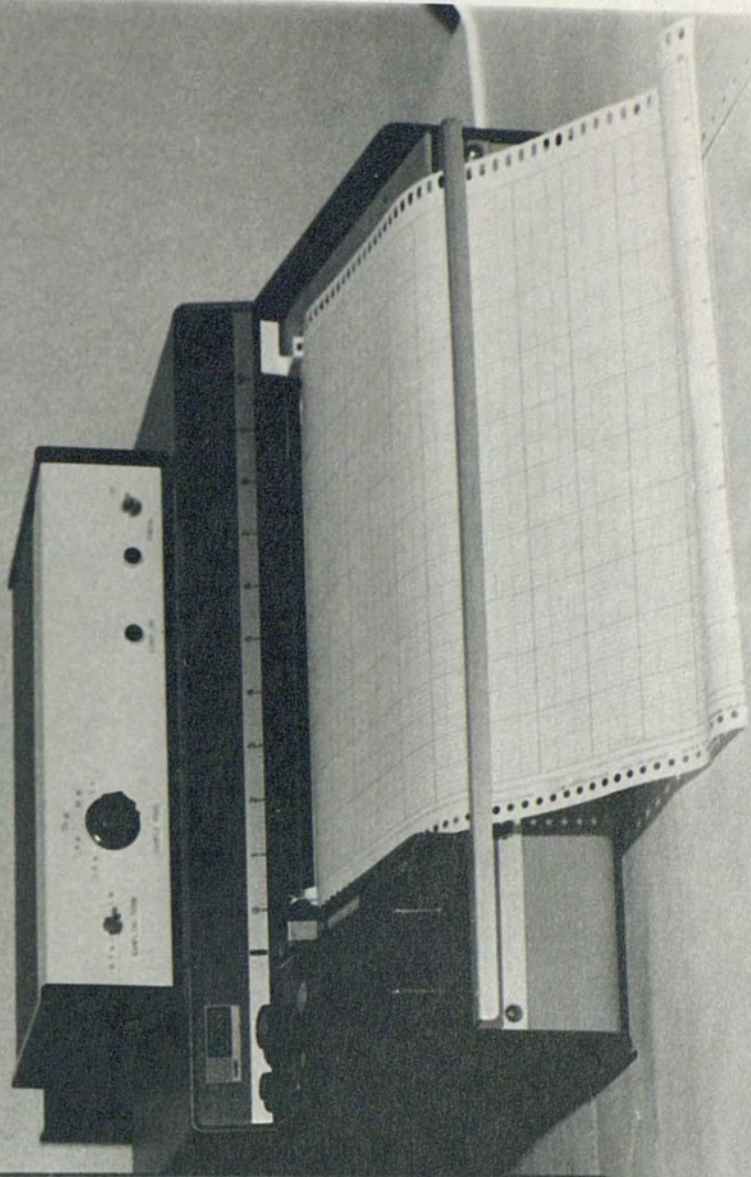
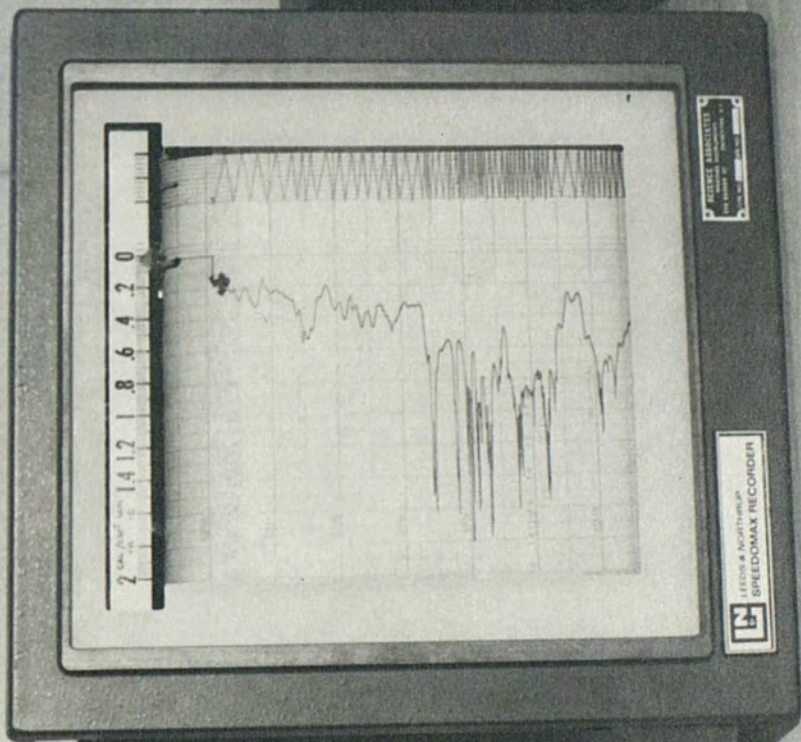


FIGURE 12a. DATA SYSTEM



FIGURE 12b. PHOTO OF DATA SYSTEM







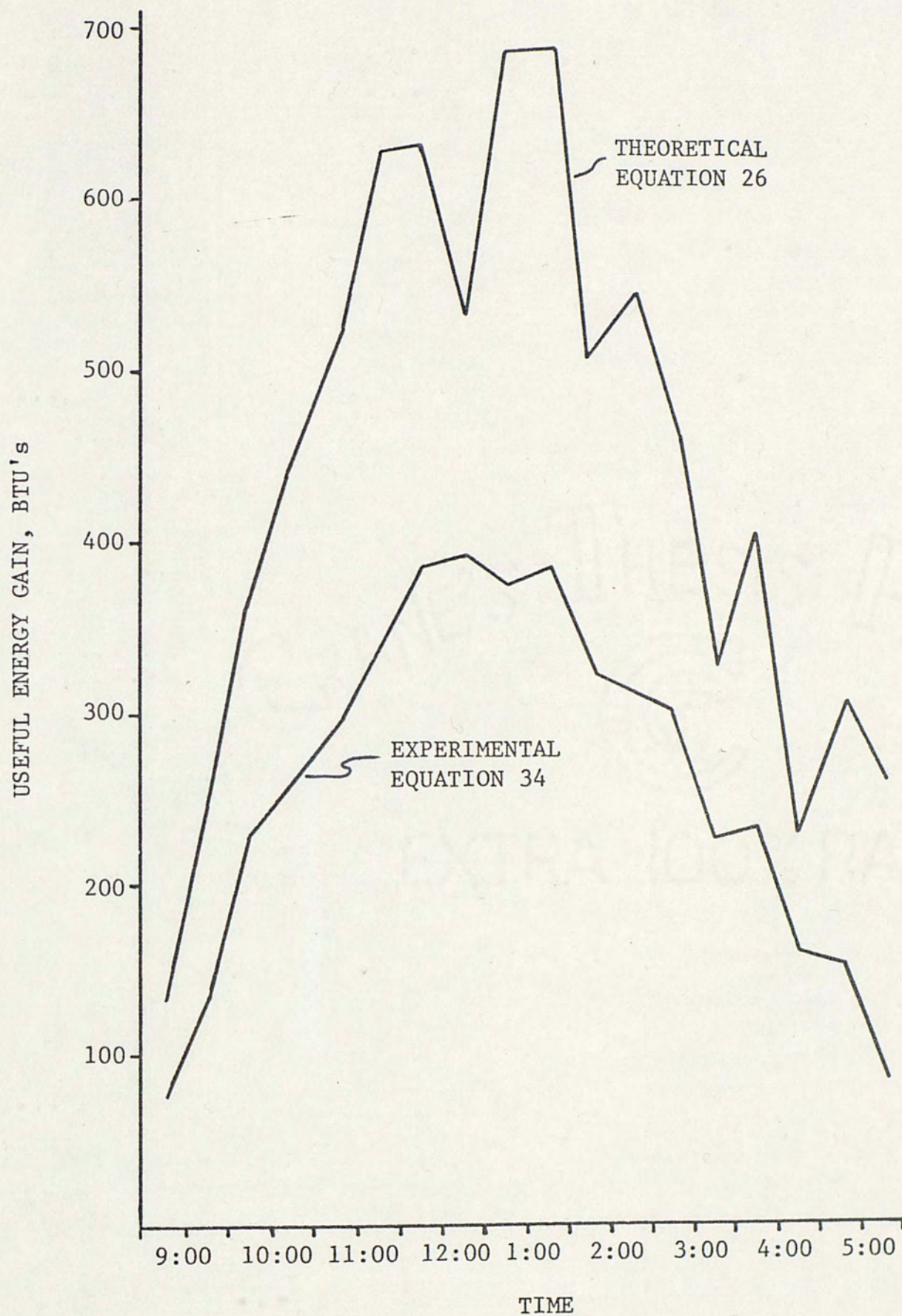


FIGURE 13. PLOT OF ACTUAL AND THEORETICAL GAIN FOR MAY 8, 1977



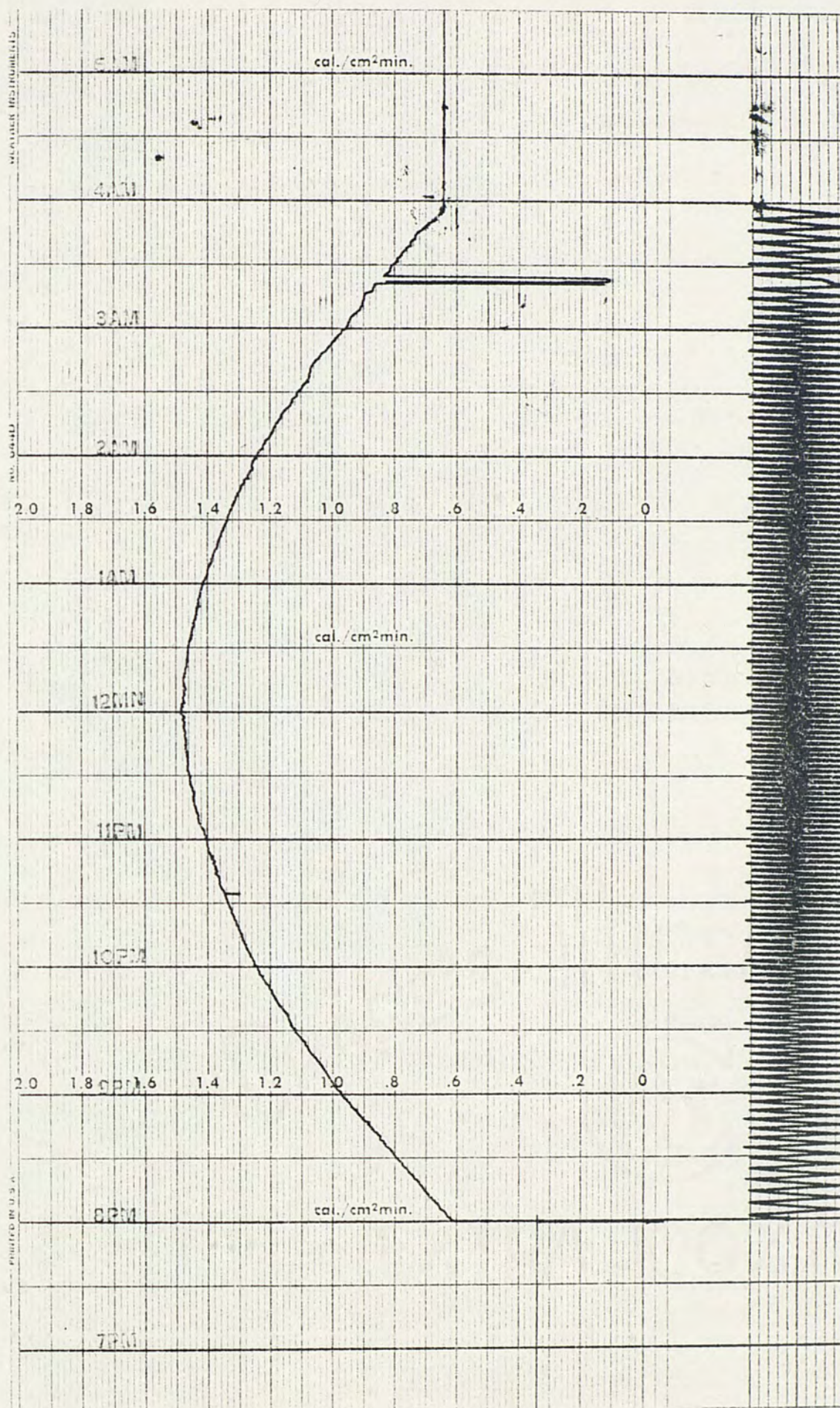


FIGURE 14. SOLAR INSOLATION FOR APRIL 8, 1977







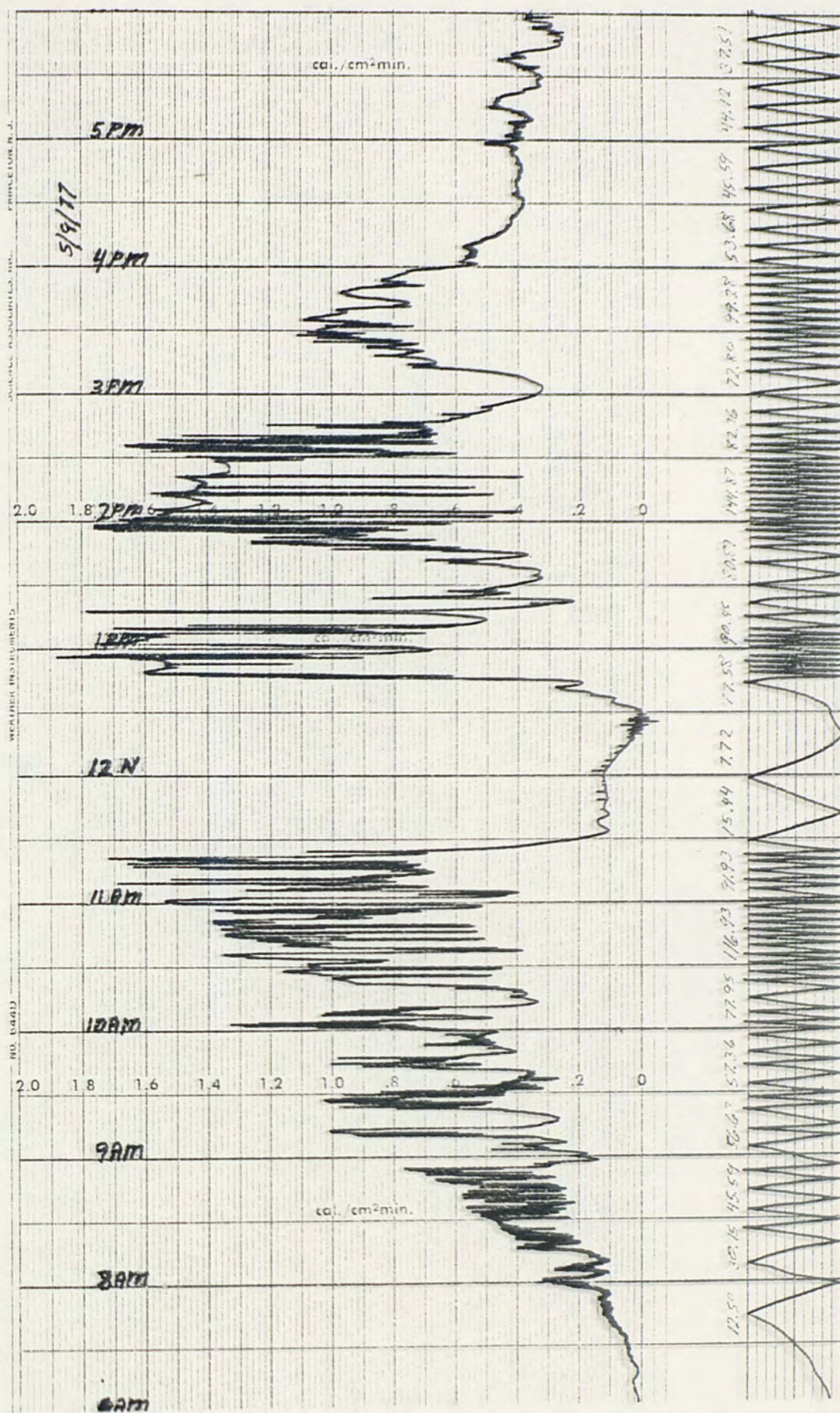


FIGURE 16. SOLAR INSOLATION FOR MAY 9, 1977



TABLE 1.  
LIST OF COLLECTOR MATERIALS

5/8" x 6' heat pipes  
.008" copper sheet metal  
3/16" double strength plate glass  
1/16" aluminum sheet metal  
Technifoam - 400 insulation  
1" x 1" x 1/16" aluminum angle stock  
2" x 2" x 1/8" aluminum angle stock  
4½" box fan



TABLE 2.

## DIODE CHARACTERISTICS

$$N = 2.0$$

$$I_r (25^\circ\text{C}) = 3 \text{ nanoamperes}$$

With these values the diode equation becomes

$$V_D = 0.052 \ln (10^9 I_D - 1)$$

$$\text{for } I_D = 1 \text{ ma}$$

$$I_o/I_r \text{ is approximately } 1.$$

therefore

$$V_D (80^\circ\text{F}) = 0.656 \text{ volts}$$

$$V_D (300^\circ\text{F}) = 0.293 \text{ volts}$$

The variation of voltage with temperature is

$$\frac{\Delta V_D}{\Delta T} = \frac{0.656 - 0.293}{220} = -1.65 \text{ mv}/^\circ\text{F}$$

$$= -2.97 \text{ mv}/^\circ\text{C}$$



TABLE 3.

## THREE DIODE TEMPERATURE SENSOR PROBE

<u>Simpson 388</u>	<u>Diode Probe</u>
- 40°C	- 39.8°C
- 30	- 30.1
- 20	- 20.0
- 10	- 9.96
0	+ 0.001
+ 10	+ 9.99
+ 20	+ 20.0
+ 30	+ 30.1
+ 40	+ 40.05
+ 50	+ 50.03
+ 60	+ 59.9
+ 70	+ 69.8
+ 80	+ 79.8
+ 90	+ 89.0
+100	+ 99.7
+110	+109.8
+120	+120.0
+130	+130.0
+140	+140.0
+150	+150.0



TABLE 4.

## PARTS LIST (EXCLUDING POWER SUPPLY)

Semiconductors

3 Quad Op-amp (U-9,10,11)	SN72LO44
2 Multiplexer	4051PC
3 Counters	CD4020AE
1 Quad NOR	CD4001
1 Quad NAND	CD4011
1 Dual Op-amp (U-14)	SN72LO22
1 Op-amp	LM741CN

Resistors

43	10K	1%
4	15K	1%
1	5K	1%
2	4.7K	5%
2	47K	5%

Capacitors

1	.01 $\mu$ fd
1	200 $\mu$ fd
1	.033 $\mu$ fd
1	.068 $\mu$ fd

Diodes

3	Diode	2N2222A
14	Diode	1N4148

Miscellaneous

1	Relay
1	Vector board, 4 x 8 inches



TABLE 5.

RECORDED COLLECTOR TEMPERATURES FOR APRIL 8, 1977

TIME	TEMPERATURE °C							(7)-(1)
	1	2	3	4	5	6	7	
8:15	23	51	40	41	21	17	21	- 2
8:45	20	62	47	46	28	19	30	20
9:15	18	72	54	53	36	20	35	17
9:45	18	80	71	69	49	22	40	22
10:15	19	83	68	66	54	24	44	25
10:45	21	90	75	72	60	26	48	27
11:15	22	97	82	78	67	28	52	30
11:45	26	101	86	81	71	33	56	30
12:15	24	103	88	83	73	31	56	32
12:45	26	102	88	82	72	32	56	30
1:15	26	99	86	80	70	33	55	29
1:45	26	96	84	78	68	33	54	28
2:15	25	92	82	76	65	31	52	27
2:45	25	88	80	73	61	32	50	25
3:15	26	82	77	68	56	34	49	23
3:45	27	73	70	62	47	33	45	18



TABLE 6.  
DATA FOR APRIL 8, 1977

TIME PERIOD	$\Delta T$ (°C)	$Q_{u,ex}$ (BTU)	I	$Q_s$ (BTU)	$\eta$
8:00 - 8:30	- 2		0.71		(-)
8:30 - 9:00	10	6.13	0.88	324.4	0.19
9:00 - 9:30	17	10.43	-1.01	372.3	0.28
9:30 - 10:00	22	13.50	1.18	435.0	0.31
10:00 - 10:30	25	15.34	1.29	475.6	0.32
10:30 - 11:00	27	16.56	1.37	505.1	0.33
11:00 - 11:30	30	18.40	1.43	527.2	0.35
11:30 - 12:00	30	18.40	1.47	541.9	0.34
12:00 - 12:30	32	19.63	1.47	541.9	0.36
12:30 - 1:00	30	18.40	1.44	530.9	0.35
1:00 - 1:30	29	17.79	1.38	508.8	0.35
1:30 - 2:00	28	17.18	1.29	475.6	0.36
2:00 - 2:30	27	16.56	1.18	435.0	0.38
2:30 - 3:00	25	15.34	1.04	383.4	0.40
3:00 - 3:30	23	14.11	0.89	328.1	0.43
3:30 - 4:00	18	11.04	0.73	269.1	0.41



TABLE 7.  
DATA FOR MAY 8, 1977

TIME PERIOD	$T_p$ (°C)	$\Delta T$ (°C)	$Q_{u,ex}$ (BTU)	$Q_s$ (BTU)	$\eta$	$Q_{u,th}$ (BTU)
8:30 - 9:00	43.67	4	73.6	375.1	0.20	133.9
9:00 - 9:30	49.67	7.33	134.9	625.1	0.22	246.6
9:30 - 10:00	59.67	12.33	226.9	882.5	0.26	356.0
10:00 - 10:30	68.67	14.33	263.7	1059.0	0.25	454.9
10:30 - 11:00	75.67	16	294.5	1184.0	0.25	524.5
11:00 - 11:30	84	18.67	343.6	1397.3	0.25	628.3
11:30 - 12:00	90.67	21	386.5	1412.0	0.27	630.4
12:00 - 12:30	90	21.33	392.5	1198.7	0.33	533.3
12:30 - 1:00	94.33	20.33	374.1	1470.8	0.25	683.1
1:00 - 1:30	85.67	21	386.5	1492.9	0.26	686.0
1:30 - 2:00	86.33	17.67	325.2	1103.1	0.29	506.6
2:00 - 2:30	85.33	17	312.9	1191.3	0.26	543.1
2:30 - 3:00	82.67	16.33	300.5	1036.9	0.29	460.9
3:00 - 3:30	72.33	12.33	226.9	742.8	0.31	326.7
3:30 - 4:00	72	12.67	233.2	934.0	0.25	407.4
4:00 - 4:30	65	8.67	159.6	558.9	0.29	228.2
4:30 - 5:00	64.33	6.67	122.8	720.7	0.17	309.8
5:00 - 5:30	61	4.67	85.9	595.7	0.14	257.4

$Q_{u,ex}$  = experimental

$Q_{u,th}$  = theoretical



TABLE 8.  
DATA FOR MAY 9, 1977

TIME PERIOD	$T_p$ (°C)	$\Delta T$ (°C)	$Q_{u,ex}$ (BTU)	$Q_s$ (BTU)	$\eta$
8:30 - 9:00	46	4.33	79.13	455.9	0.17
9:00 - 9:30	48.67	5.33	98.1	566.3	0.17
9:30 - 10:00	51.67	7.33	134.9	573.6	0.24
10:00 - 10:30	38	11.67	214.7	779.5	0.28
10:30 - 11:00	70.33	16	294.5	1169.3	0.25
11:00 - 11:30	73	14.67	270	919.3	0.29
11:30 - 12:00	55	5.67	104.4	154.4	0.68
12:00 - 12:30	38	3.33	61.3	77.2	0.79
12:30 - 1:00	47.67	9	165.6	775.8	0.21
1:00 - 1:30	65.33	15	276	904.5	0.31
1:30 - 2:00	65	13.67	251.6	808.9	0.31
2:00 - 2:30	86	23	423.3	1448.7	0.29
2:30 - 3:00	68	14.33	263.7	823.6	0.32
3:00 - 3:30	63	11.67	214.8	728.0	0.30
3:30 - 4:00	69	15.33	282.1	992.8	0.28
4:00 - 4:30	56	9	165.6	536.8	0.31
4:30 - 5:00	51.67	6.67	122.8	455.9	0.27
5:00 - 5:30	50.67	5.67	104.4	441.2	0.24



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